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DONALDSON'S

PONCELET TURBINE

AND

WATER-PRESSURE ENGINE AND PUMP

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DONALDSON'S
PONCELET TURBINE
AND
WATER-PRESSURE ENGINE AND PUMP.

PREFACED BY A SHORT TREATISE
ON THE
IMPULSIVE ACTION OF INELASTIC FLUIDS.

IS THE RELATIVE VELOCITY AFTER IMPACT EQUAL TO THE RELATIVE VELOCITY BEFORE IMPACT?

BY

WILLIAM DONALDSON, M.A., M.I.C.E.,

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INTRODUCTION.

In a work published in 1876, entitled 'Principles of Construction and Efficiency of Water-wheels,' the author of the present pamphlet argued against the dogma which maintains that the relative velocity after impact is equal to the relative velocity before impact. This dogma is in reality the foundation of the belief that Turbine efficiency can reach nearly 80 per cent. of the gross power.

In the work referred to the author gave a mathematical demonstration that the doctrine of equality is untenable, and showed that Professor Rankine had, through basing his calculations on this doctrine of equality, arrived at a general formula for the work done by a jet striking a vane, which is utterly incorrect, except in cases unaffected by the correctness or incorrectness of the theory.

The author also endeavoured to show that the doctrine that the statical measure of the impulsive pressure of a jet against a vane is double the pressure due to the head producing the velocity of the jet is incorrect.

In the present pamphlet the same views are advocated with as little reference as possible to mathematical formulæ. Those who may by this pamphlet be induced to take an interest in the question will find the subject fully discussed in the treatise on Water-wheels.

At the time of the publication of the treatise on the 'Principles of Construction and Efficiency of Water-wheels,' very meagre criticisms were given in the engineering journals. No notice was taken of the keystone of the whole work—Is the relative velocity after impact equal or not equal to the relative velocity before impact?

The author hopes that the editors of the engineering papers, which exert such great influence on the engineering world, will not again miss the opportunity afforded them by the publication of the present pamphlet.

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DONALDSON'S
PONCELET TURBINE
AND
WATER-PRESSURE ENGINE AND PUMP.

CHAPTER I.

ON THE IMPULSIVE ACTION OF INELASTIC FLUIDS.

IF the doctrine that the relative velocity, leaving friction out of consideration, after impact is equal to the relative velocity before impact is correct, it follows that if the vane is at rest the absolute velocity is not altered by the impact, or, in other words, that the *vis viva* of a mass of water striking a vane at rest, either normally or otherwise, is unaffected by the impact. A favourite question with the advocates of the theory of equality is this:—If the *vis viva* after impact is not equal to the *vis viva* before impact, what has become of the difference? The next step on this process of reasoning is naturally the doctrine that the *vis viva* lost in passing through a turbine is accounted for exactly by the external work done by the turbine in the interval.

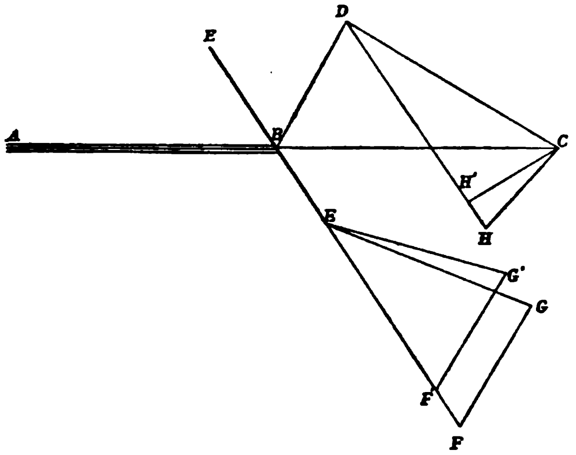
In short, according to this theory, the power lost by the water in passing through turbines, leaving loss by friction alone out of consideration, is equal to the power imparted to the vanes.

An analogous case would be that of two inelastic masses $m\,m'$, impinging against one another. Suppose, for the sake of simplicity, that the mass m' is at rest before impact. If the mass m be moving with a velocity v before impact its capacity for work will be equal to $\frac{m\,v^2}{2}$. After impact the two masses move on in contact with a common velocity v, which is equal to $\frac{m\,v}{m+m'}$, so that the sum of the capacity for work finally possessed by the masses m and m' will be equal to $\frac{m^2\,v^2}{2\,(m+m')}$, which is always less than the original capacity possessed by m, and may be indefinitely small. If the balls are elastic and of the same substance, whose modulus of elasticity is e, we shall have the velocity of the mass m after impact equal to $\frac{(m - e\,m')\,v}{m + m'}$ and that of m' equal to $\frac{(1 + e)\,m\,v}{m + m'}$, so that the final capacity for work is equal to $\frac{v^2 m\,(m + e^2 m')}{2\,(m + m')}$, which in one case only is equal to $\frac{m\,v^2}{2}$, viz. when the elasticity is perfect.

It may be that the principle, properly interpreted, is correct, and that the power lost by the first ball is partly expended in communicating momentum to the second, and partly in producing heat, which in the case of perfectly elastic balls is at once given back again in the shape of an increase of the momentum due to direct impact, and in the case of inelastic balls remains stored up, and simply increases the temperature of the masses m and m'. Any discussion as to the relative temperatures of the water and the vane after and before impact is, however, irrelevant. The author will have gained his point if he can prove, either by argument or experimental results, that conclusions as to the efficiency of water-power machines, based on the assumption, to use Viry's own words, "rien se perd dans la nature," must be erroneous, when that maxim is interpreted to mean that the effective power communicated to the machine is equal to the power lost by the water.

If tangential friction be left out of consideration, the incorrectness of the theory that the *vis viva* after impact is equal to the *vis viva* before impact may be demonstrated as follows:—

Fig. 1.



Let the jet strike the vane B E at the point B, and let B C, B D represent the initial velocities of the jet and vane respectively in magnitude and direction. Join D C.

D C will represent in magnitude and direction the motion of the jet relatively to the vane before impact.

Draw E F parallel to the face of the vane and equal to D C; also through F draw F G parallel and equal to B D, and join E G. Through D draw D H parallel and equal to E F, and join H C. If, therefore, the relative velocity of the water and the vane after impact be equal to the relative velocity before impact, E G will represent in magnitude and direction the final absolute velocity of the water, and H C the direction and magnitude of the change of motion of the jet during the interval of contact with the vane. Also D H is equal to D C and the two angles D H C, D C H, are equal to each other, therefore each must be less than a right angle, and therefore the direction C H, in which the jet has been deflected, is not at right angles to D H, which is parallel to the surface of the vane.

Therefore the change of velocity caused by impulse against a plane vane has been in a direction other than normal to the vane, friction being left out of consideration, which is impossible. Therefore the relative velocity after impact cannot be equal to the relative velocity before impact. Draw C H′ at right angles to D H. In like manner it may be shown that no other line than C H′ can represent in magnitude and direction the change of the motion of the jet, therefore the relative velocity after impact will be represented by the line D H′. In E F take the point F′, such that E F′ = D H′, and draw F′ G′ parallel and equal to B D. Join E G′. Then will E G′ represent in magnitude and direction the absolute velocity of the jet after leaving the vane.

In the Poncelet wheel the tangent to the upper edge of the curve made by a vertical section of the vane is radial, and the tangent to lowest point parallel to the relative velocity of the water and the vane.

Of these wheels Professor Rankine remarks that the water finally glances off the vane in a direction relatively to the vane, parallel, but opposite to that in which it glanced on to the vane, so that the final absolute velocity will be radial, if we look upon the vane's motion in the interval as wholly linear at right angles to the radius through the point of impact. Therefore, in accordance with his rule that "the energy exerted by the water on the vane (*in direction of motion*) is equal to the energy lost by the water," the efficiency will be equal to $\cos.^2 a$, where a is the angle between the direction of the jet and the vane, friction being left out of consideration. The words in italics are added by the author. They are necessary to make the statement a faithful interpretation of the equation, (see Rankine's Prime Movers)

$$\mathrm{P}\,u = \frac{\mathrm{W}\,(v_1^2 - v_2^2)}{2g}.$$

This is the accepted theory of the Poncelet wheel. To illustrate it more clearly, let us consider the action of an indefinitely small particle of water of mass $\delta\, m$, and in order to simplify the action of gravity, let us suppose the motion of the vane horizontal. Let a velocity $\frac{v_1 \cos a}{2}$ be impressed on both the vane and the particle in a direction opposite to that of motion so as to bring the vane to rest. If neither friction nor curvature cause loss of velocity, the particle will rise to the height due to the relative velocity, and at that instant its relative velocity will be nil and absolute velocity equal to $\frac{v_1 \cos a}{2}$.

If then $c\, v_1^2$ represent the energy communicated to the vane, we shall have, in accordance with the theory just enunciated, since the energy lost by the water has been exerted in imparting energy to the vane and overcoming the resistance of gravity,

$$c v_1^2 + \delta m g \mathrm{H} = \frac{\delta m v_1^2\left(1 - \frac{\cos^2 a}{4}\right)}{2};$$

where H is the height due to the relative velocity. The particle will then begin to descend, and its relative velocity, when it reaches the bottom of the vane, will be equal to the initial relative velocity in magnitude, but will be opposite to it in direction, so that the final absolute velocity will be simply v_1 sin. a vertical, and if $c' v_1^2$ represent the energy imparted to the vane during the descent we shall have:—

$$c' v_1^2 - \delta m g \mathrm{H} = \frac{\delta m v_1^2\left(\frac{\cos^2 a}{4} - \sin^2 a\right)}{2};$$

therefore,

$$(c + c') v_1^2 = \frac{\delta m v_1^2 \cos^2 a}{2}.$$

If for H we substitute its value—

$$\frac{v_1^2\left(\frac{\cos^2 a}{4} + \sin^2 a\right)}{2 g},$$

we get

$$c v_1^2 = c' v_1^2 = \frac{\delta m \cos^2 a}{4};$$

so that the energy imparted during the ascent has been equal to the energy imparted during the descent. The whole energy imparted will therefore, theoretically, be equal the sum of the energies exerted by the separate particles; but when we consider the action of the whole together, it is evident that only those particles which strike the vane first will reach the top, the subsequent particles being obstructed in their ascent by the return of the first, and the particles which first reach the top will not act on the vane on their return, being forced from it by the upward motion of the succeeding ones. So that the energy imparted to the vane, due to both direct action and reaction, will be much less than that assigned by theory in the case of one particle.

In order to eliminate the effect of the action of gravity, we will suppose that the axis of the wheel is vertical, so that the motion of the jet is horizontal. At the point where the tangent to the vane is radial, since the relative velocity is unaffected by the impulse, the absolute velocity will be equal to

$$v_1\sqrt{\frac{\cos^2 a}{2} + \sin^2 a},$$

and therefore the energy lost by the water and imparted to the vane will be equal to $\frac{\delta m v_1^2 \cos^2 a}{4}$. Imagine now another Poncelet vane added to this, with its radial end meeting this tangentially. The relative velocity when it reaches the other end of the vane will be unaltered in magnitude, but opposite in direction, so that the final absolute velocity will be equal to $v_1 \sin a$, and therefore the energy lost by the particle of water during contact with the second half of the vane will also be equal to $\frac{\delta m v_1^2 \cos^2 a}{4}$. Since the flow is continuous, and the water discharged freely from the opposite end of the vane, one particle of water does not interfere with another, and the efficiency of such a compound vane is theoretically equal to $\cos.^2 a.$.

The vanes of the Poncelet Turbine are designed so as to comply strictly with Poncelet's theory, and therefore the theoretical coefficient of efficiency is equal to $\cos.^2 a$.

Owing to the circular motion of the vanes in the case of a Turbine, the two parts of the compound vane will be of a totally different shape. At the point at which they meet one another, the tangent to the relative path will coincide with a radius of the wheel. By varying the value of the ratio $\omega R_1 \div v_1 \cos. a$, the position of this point of junction may be made to vary *ad libitum.* If the value $\omega R = v_1 \cos. a$ is selected, the first half of the compound vane disappears altogether, and the second half cuts the receiving side of the wheel at right angles. The less the value $\omega R_1 \div v_1 \cos. a$ the greater will be the length of the first half of the vane. This first half cannot, however, be made equal to the theoretical best length of the vane comprised between the radii R_1 and R_0 (see Chapter II.). Corresponding with the numerical values given in figures (2) and (3), Chapter II., the position of this point is as follows:—

(1) In outward-flow turbines when $\omega R_1 \div v_1 = \cdot 34\, v_1$
Distance of junction of vanes from centre $= 1 \cdot 17\, R_1$.

(2) In inward-flow turbines when $\omega R \div v_1 = \cdot 53\, v_1$
Distance of junction of vanes from centre $= \cdot 86\, R_1$.

The mass of the volume of water discharged through an orifice whose area is a, with a velocity v, is equal to $\frac{W\,a\,v}{g}$, if W be the weight of an unit of volume. If a flat vane be held at right angles to the jet, the resistance R against the vane, caused by the deflecting of the jet, will be equal to $\frac{W\,a\,v^2}{g}$. If now the vane be pressed close to the orifice, so as to stop the flow of the water, the theoretical resistance against the vane will be equal to $\frac{W\,a\,v^2}{2\,g}$, because the theoretical height due to the velocity v is equal to $\frac{v^2}{2\,g}$. Some writers on fluid impact have deduced from these expressions the astonishing conclusion that the pressure exerted by a jet against a vane is equal to double the statical pressure due to the theoretical head producing the velocity. It is a case, however, of comparing unlike things. In the case of the jet, R is the resistance overcome in an unit of time, and is equal to the instantaneous impulse with velocity v of a mass equal to that of the fluid discharged in an unit of time, and does not admit of comparison with the statical pressure due to the head producing the velocity.

CHAPTER II.

THE PONCELET TURBINE.

In this chapter the form of the vanes and guide-blades, and the best proportions of the various parts only, are given. The mechanical contrivances for starting, stopping, and regulating the flow of water in each case, are of the same kind as those used for other turbines of the same class.

The following symbols will be used in the formulæ :—

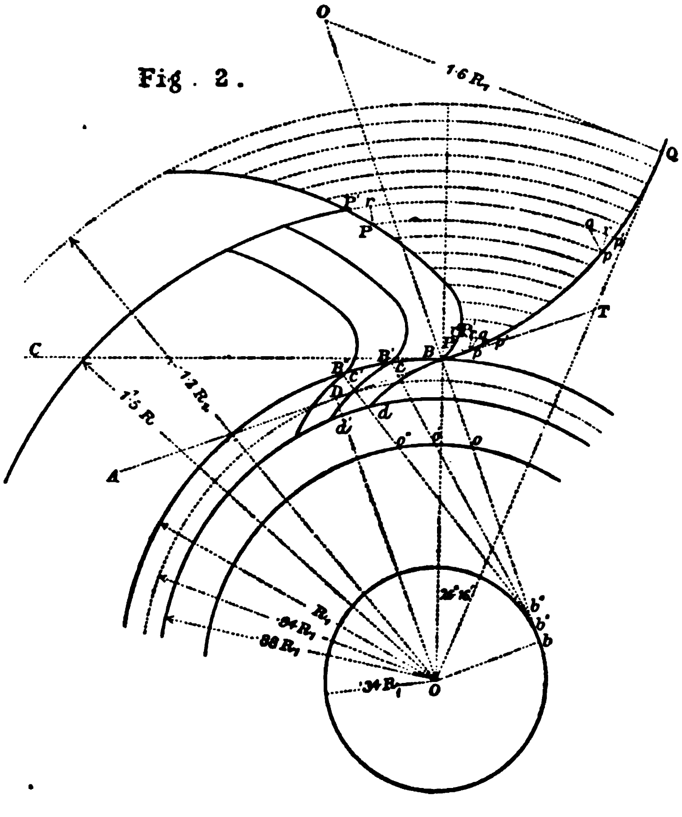
v_1 = velocity of discharge from guide-blades in feet per second.
a = angle between direction of jet and direction of motion.
δ = „ radii through the point of impact and the point where the absolute path cuts the discharging side.
R_1 = radius of receiving side of wheel.
R_0 = „ discharging „
ρ = any intermediate radius between R_1 and R_0.
ω = angular velocity of wheel.
D = depth of guide-blade chamber.
r = radius of curve of path of absolute velocity.

The numerical ratios given in the diagrams which give the proportions of the various parts in terms of R_1 are theoretically the best. Where they depend upon the value of a, this has been taken equal to 20°. The less a is, the greater will be the efficiency; but for values of a less than 20° the increase is very slight. Thus for the value $a = 10°$, whilst the discharging capacity is reduced by nearly one-half, the efficiency is only increased in the ratio 97 : 89 ; in the very best turbines the increase is less than 5 per cent., an increase, in all probability, more than counterbalanced by the work done in overcoming the increase in frictional resistance of the external water alone due to the increased speed.

Outward-flow Turbine.

Fig. 2 is a diagram of the vanes and guide-blades of an outward-flow Turbine.

Fig. 2.



Curve of Guide-blades.

The tangents to the guide-blades at the points where these cut the periphery are all tangents to the circle radius O D = $R_1 \cos. a$, and the normals to the guide-blades through the same points of intersection are all tangents to the curve radius O b = $R_1 \sin. a$. Let b be the point where one of these normals meets the circle radius O b. From b measure off distances $b\,b'$, $b'\,b''$, which are each equal to B B′ $\sin. a =$

$B'B''$ sin. a = &c., where BB', $B'B''$, &c., are the distances between the centres of successive guide-blades measured along the circumference of the circle radius R_1. The internal radius of the guide-blade chamber must be made equal to R_1 (2 cos. a − 1), so that the circle radius O D is exactly in the middle of the guide-blade chamber. Describe another circle $o\,o'\,o''$ with a radius equal to about two-thirds of the radius of the inner side of the guide-blade chamber. Join b B, b' B′, b'' B″, &c., meeting this last circle in o, o', o'', &c. With centre b and radius $b\,B = O\,D = R_1$ cos. a describe the arc B c, meeting the next succeeding straight line B′ b' in c, with centre o' and radius $o'\,c$ describe the arc $c\,d$ meeting the inner circumference of the guide-blade chamber in d. The curve B c d will be a suitable curve for the guide-blade. Similarly describe the curve B′ c' d', starting with centre b' and so on for the rest.

Curve of Vanes.

In order to describe the best form of vane, that value of the ratio $R_0 \div R_1$ must be adopted which gives the form of vane of maximum efficiency. In the Figure 2, $OB = R_1$ and $OQ = R_0$. The ratio $R_0 \div R_1$ ought not to be taken less than 2. This value makes the width of the theoretically best wheel too great for actual use, and it must therefore only be used to design the shape of the vane.

In the diagram the external radius of the wheel is taken equal to $1{\cdot}5\,R_1$ only. The best theoretical velocity of the periphery of the circle radius R_1 is equal to about $\cdot 32\,v_1$. The height due to the absolute velocity at the point where $\rho = 1{\cdot}46\,R_1$ is equal to $\frac{\cdot 16\,v_1^2}{2g}$, friction being left out of consideration. For the value $\rho = R_0 = 2\,R_1$, the height due to the absolute velocity, is equal to $\frac{\cdot 11\,v_1^2}{2g}$, or, in other words, cutting off the width at the radius $\rho = 1{\cdot}5\,R_1$, leaves about 5 per cent. of the whole head not utilised. As only a fraction of this 5 per cent. would in reality be utilised, the saving in friction much more than counterbalances the loss.

The following formulæ give the values of the element of the curve of the vane:—

$$\delta = \angle\, BOQ = 2 \tan^{-1} \frac{1 - \tan\frac{a}{2}}{3\left(1 + \tan\frac{a}{2}\right)}.$$

$$r = O'Q = O'B = \frac{R_1 \sin\delta}{1 - \sin(a + \delta)}.$$

$$pP = \frac{\omega\rho(\rho - R_1)}{v_1 \sin a}.$$

where $\rho = OP = Op$.

The best velocity of the periphery is given by the relation $\omega R_1 = \cdot 32\, v_1$ when $\alpha = 20°$, and the above equation becomes for these values

$$pP = \frac{\rho(\rho - R_1)}{R_1},$$

$$\delta = 26° \, 16' \, 30'',$$

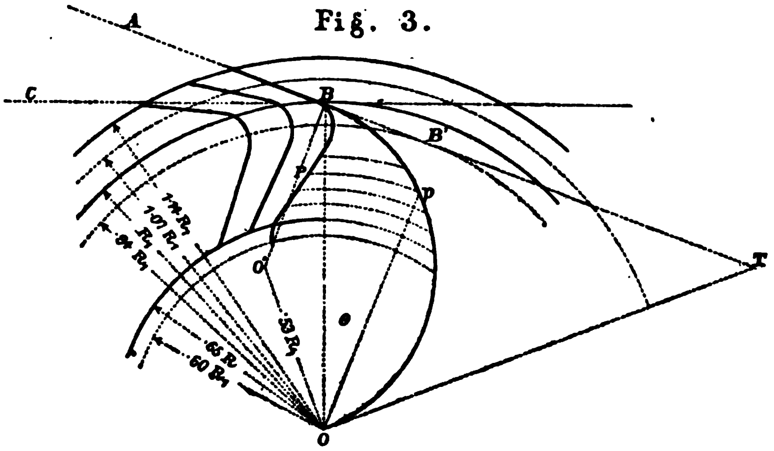
$$r = 1 \cdot 6\, R_1.$$

The depth of the wheel ought to be the same throughout, and equal to that of the guide-blade chamber. This last depth ought never to exceed one-third of the value of R_1, the inner radius of the wheel.

Inward-flow Turbines.

Fig. 3 is a diagram of the curves of the guide-blades and vanes of an inward-flow Turbine.

Fig. 3.



Shape of Guide-blades.

In the figure the guide-blades are simple flat vanes which all touch a cylinder whose radius is equal to $R_1 \cos. \alpha$. A set of guide-blades of the same design as those used in Professor Thompson's vortex Turbine answer best where the water brings along much solid matter. Where the water is effectually screened those shown in the figure are the most efficient.

Shape of Vanes.

The best curve of absolute velocity is one which passes through the centre of the wheel and cuts the external periphery at an angle equal to a. The radius of this curve is given by the equation

$$r = \mathrm{O'B} = \mathrm{O'O} = \frac{\mathrm{R}_1 \sec a}{2}.$$

$$\delta = \mathrm{BOT} \qquad = \frac{\pi}{2} - a.$$

For the reasons which led the writer to adopt $\mathrm{R}_0 = \cdot 65\,\mathrm{R}_1$, the reader is referred to the work on water-wheels.

The length of the various arcs Pp are given by the equation

$$\mathrm{P}p = \frac{(\mathrm{R}_1 - \rho \cos \theta)\, \omega \rho)}{v_1 \sin a},$$

where θ is equal to the angle between the initial radius O B and the radius O p.

The value of $\omega \mathrm{R}_1$ ought not to exceed $\cdot 53\, v_1$, and probably the best value is given by the equation

$$\omega \mathrm{R}_1 = \cdot 5\, v_1 \cos a = \cdot 47\, v_1\,;$$

with this value we get

$$\mathrm{P}p = \frac{1 \cdot 4\, \rho\, (\mathrm{R}_1 - \rho \cos \theta)}{\mathrm{R}_1} \text{ nearly.}$$

also $\delta = 70°. \qquad r = \cdot 53\, \mathrm{R}_1.$

The depth of the wheel ought to go on increasing from the outer to the inner circumference in a ratio somewhat greater than that of the ratio $\mathrm{R}_1 \div \rho$. If they are exactly in that ratio, the area perpendicular to the direction of the velocity of discharge v_1 sin. a will be constant, but no allowance will have been made for friction. In small turbines it will be necessary to make a larger allowance for friction than in large ones. No loss of efficiency, except through increased outside surface friction, will be experienced by making this depth somewhat greater than is necessary, nor will any diminution of efficiency arise from making the wheel with parallel sides, providing the external depth exceeds the depth of the guide-blade chamber multiplied by the ratio $R_1 \times R_0$ sufficiently to allow for frictional retardation of velocity. If we add one-quarter to the depth given by the equation

$$\text{Depth of wheel} = \frac{R_1 D_1}{R_0}$$

for small wheels and one-eighth for large wheels, we get,

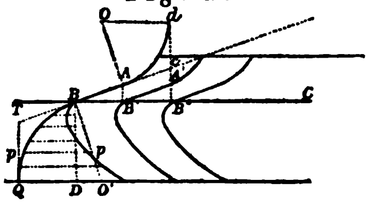
$$\text{Depth of wheel} = 2\,D \text{ for small wheels}$$
$$= 1 \cdot 7\,D \text{ for large wheels.}$$

D being the depth of the guide-blade chamber.

Mr. Morton, of the Errol works, has made several of these wheels.

Parallel-flow Turbine.

Fig. 4.



In parallel-flow Turbines there are three difficulties to contend with which do not occur in the case of outward and inward-flow Turbines. Unless they are of uniform depth, the radius of the absolute path which is given by the equation

$$r = D \sec \alpha$$

will vary, and since the relative path is obtained from the equation

$$p\,P = \frac{B\,b\,\omega\,\rho}{v_1 \sin \alpha},$$

the shape of this relative path, which is the curve of intersection of concentric cylinders with the vane, will vary at each section. Also the ratio $\omega\,\rho \div v_1$ will vary for all values of ρ from R_0 to R_1. The only way to get over these difficulties is to make the ratio $R_0 \div R_1$ very small. The writer cannot recommend the use of parallel flow-Turbines. Those who wish to see the question fully discussed are referred to the work on water-wheels.

CHAPTER III.

WATER-PRESSURE ENGINE AND PUMP.

The machine, whether used for power or pumping, consists of two or more cylinders enclosed in a water-tight case. The cylinders are attached in a row to one side of the case by means of flanges, and one end of each cylinder projects beyond the side to which they are attached. The length of this projection is equal to the sum of the depths of the piston and of a set of portholes. The depth of these portholes is equal to the clearance between the side of the casing out of which the cylinders project and the piston, when this is at the extreme point of the stroke.

Both ends of the cylinder are open, but the end of the cylinder inside the case is provided with a lid which is kept in position by the pressure of the water in the case by which all the cylinders are surrounded. Portholes are provided in both water-pressure engine and pump close to the commencement of the stroke. These portholes are opened and closed by a cylindrical valve, which is actuated by one or two excentrics attached to the crank shaft.

The above description applies to both the power engine, and the pump. In the case of the water-pressure engine the space between the cylinders and the casing is kept filled with water under any assigned pressure. In the case of the pump the water raised is discharged through the portholes and this outside casing.

Water-pressure Engine.

In the engine there is a third set of portholes opposite the cylinder lid. The lid of the cylinder also contains portholes opposite to these. During the return stroke of the piston the valve portholes are closed and only begin to open just at the commencement of the stroke. No water, therefore, which may have got into the cylinder can escape through the portholes, which are opened and closed by the valve. The contained water, however, raises the lid sufficiently high to admit of its escape through the third set of portholes. The portholes in the lid are provided to guard against the possibility of the valve getting jammed in a wrong position opposite the third set of cylinder portholes.

The lid must also be provided with air-vales to allow of the admission of the air when the water is being discharged by gravitation out of the portholes at the open end of the cylinder, and also to admit of the escape of the air during the return stroke.

If two excentrics are used, link motion may be applied to alter the length of the stroke, stop the engine, and reverse the direction of rotation. When only one excentric is used, the cylindrical valve must be made of a length to suit the shortest working stroke.

To the valve is attached a balanced weight W, so adjusted that the centre of gravity of the valve and the weight are situated in the centre of this rod.

The cylinders may be placed at any inclination, but by preference they ought to be vertical. When they are vertical the moving parts exactly balance each other, and the only force consumed internally is that due to the friction on the bearings and the slide bars. No packing is used either with the cylinder piston or the piston-rods. The piston-rods are always in tension whatever may be the inclination of the cylinders; when these are vertical, single brasses only are required for the bearings.

Sheet No 1 gives the details of a four-cylinder engine, each cylinder being 9½ inches in diameter. Under a 400-foot head at 40 revolutions a minute the engine will do fully 110 horse-power.

On Sheet No. 2 is shown the mode of using two excentrics with link motion. The lengths of the rods and the bar of the link are so adjusted that the bar itself is straight.

The maximum number of revolutions which water-pressure engines of various sizes of cylinders can run per minute may be calculated as follows.

The engine shown on Sheet No. 1 has a maximum working stroke equal to $\frac{5}{6}$ths of the full stroke, or the emptying of the cylinder by gravitation takes place during $\frac{1}{6}$th of the stroke.

This corresponds with a little over 96°, so that if n be the number of strokes per minute the whole interval of emptying will be equal to

$$\frac{96}{360} \times \frac{60}{n} = \frac{16}{n} \text{ seconds.}$$

Now, in describing the first half of the porthole opening, which will be equal to $\frac{1}{12}$th of the full stroke, the crank radius will pass through an angle of 15° only, and through an angle of 33° in the second half, so that the time occupied in the first stage is equal to $\frac{15}{48} \times \frac{8}{n} = \frac{15}{6n}$ seconds, and during the second half to $\frac{33}{48} \times \frac{8}{n} = \frac{33}{6n}$ seconds. Since the second half of each of above intervals will be described more slowly than the first half, we shall be on the safe side if we take the area of the outlet to be equal to the $\frac{1}{4}$th and $\frac{3}{4}$ths of the whole opening during the respective intervals, or equal to

$$\frac{\pi D - w}{24} L \text{ and } \frac{\pi D - w}{8} L,$$

where D is the diameter of the cylinder in feet, L the full length, and w the width of the bars of the portholes. Since these bars have only to act as guides, the total width need never exceed $\frac{1}{8}$th, and in the case of large cylinders may be much less than $\frac{1}{8}$th of the whole circumference. Hence if v be the average velocity of discharge, in order that the working barrel may be empty before the return stroke has traversed one-half of the porthole openings we must have

$$\frac{7\pi \mathrm{D}\mathrm{L}v}{8n}\left\{\frac{1}{24}\times\frac{15}{6}+\frac{1}{8}\times\frac{33}{3}\right\}=\frac{\pi \mathrm{D}^2}{4}\times\frac{5\mathrm{L}}{6},$$

whence $v = \cdot 16\, n\, \mathrm{D}$.

Now the head due to this velocity is equal $\frac{v^2}{2gc^2}$, where c is the coefficient of discharge, which for openings of this size close to the bottom of the vessel we may take to be certainly not less than $\cdot 7$, and with this value of the coefficient we get

$$\mathrm{H} = \cdot 0008\, n^2 \mathrm{D}^2.$$

The height of the working barrels of the cylinders is about three times the diameter, and we shall be therefore on the safe side if we assume that the height due to the average velocity may be equal to one diameter, since the maximum depth of the porthole is equal to about one-half of the diameter. With this maximum value of H, the maximum number of revolutions will be given by the equation

$$n = \frac{36}{\sqrt{\mathrm{D}}}.$$

If the working stroke is equal to $\frac{7}{8}$ths of the full stroke, the angular interval of discharge will be equal to a little less than 83°, and the whole time for emptying will be equal to $\frac{83}{360}\times\frac{60}{n}=\frac{14}{n}$ seconds nearly, and the crank radius will describe angles of about 13° and 29°, whilst the last but one and last sixteenth of the stroke is being done, and we get the equation

$$\frac{7\pi \mathrm{D}\mathrm{L}v}{8n}\left\{\frac{1}{32}\times\frac{13}{6}+\frac{3}{32}\times\frac{29}{3}\right\}=\frac{\pi \mathrm{D}^2}{4}\times\frac{7\mathrm{L}}{8};$$

whence

$$v = \cdot 26\, n\, \mathrm{D}$$

$$\mathrm{H} = \cdot 002\, n^2 \mathrm{D}^2.$$

For the maximum value H = D we get

$$n = \frac{22}{\sqrt{D}}.$$

When the working stroke is equal to $\frac{9}{10}$ths of full stroke,

$$n = \frac{15}{\sqrt{D}}.$$

The following table gives the maximum number of revolutions calculated from these formulæ :—

TABLE I.

Diameter of Cylinder in Inches.	Maximum Number of Revolutions.		
	Working Stroke = $\frac{5}{8}$ths Full Stroke.	Working Stroke = $\frac{7}{8}$ths Full Stroke.	Working Stroke = $\frac{9}{10}$ths Full Stroke.
4	64	40	27
6	50	31	21
8	44	27	18
10	39	24	16
12	36	22	15
15	32	20	13
18	29	18	12
21	27	16	11
24	25	15	10
30	22	14	9
36	20	12	8

The theoretical efficiency of the water-pressure engine may be calculated in the following way :—

The loss of head is due to four causes :—

(1) Friction in supply pipe.
(2) Clearance for tail water.
(3) Head due to velocity of water in cylinder.
(4) Head lost by friction and contraction in passing inlet valve.

No 1 is common to all machines.

No. 2, leaving out of consideration the work done by the water whilst the cylinder is emptying, will be equal to the height of the middle full stroke above the surface of the tail water.

No. 3. The velocity will vary with the length of the stroke and the number of revolutions per second.

The following table gives the head due to velocity in a 3-foot cylinder for ten revolutions per minute, the working stroke being $\frac{5}{6}$ths of the full stroke, which is equal to 7 feet 10 inches:—

TABLE II.

Spaces each equal to $\frac{1}{6}$th Stroke.	Degrees swept out.	Time of Passing each Space in Seconds.	Velocity in Inches.	Height due to Velocity in Feet.
1	48	·8	19·6	·04
2	22	·37	42·4	·20
3	20	·33	47·6	·24
4	20	·33	47·6	·24
5	22	·37	42·4	·20

Average height due to velocity ·19.

Since the length of the full stroke is the same whatever may be the working stroke, the mean head due to velocity when the working stroke is equal to $\frac{7}{8}$ths or $\frac{9}{10}$ths of the full stroke will be slightly less than this, because the final velocity will be less. For the present purpose the mean head may, however, be taken to be the same in each case.

The loss due to friction in the cylinder itself is inappreciable. The two next tables give the heights due to velocity through the portholes for a cylinder of the same size and for the same number of revolutions: Table III., when the working stroke is equal to $\frac{5}{6}$ths of the full stroke; Table IV., when the working stroke is equal to $\frac{7}{8}$ths of the full stroke. The 1st, 2nd, 3rd, and 4th spaces in both tables are equal to $\frac{1}{6}$th of the full stroke. In Table III. the 5th is equal to $\frac{1}{12}$th and in Table IV. to $\frac{1}{8}$ of the full stroke, whilst the 6th and 7th are each taken equal to $\frac{1}{24}$th of the full stroke, for the purpose of showing the effects of wire drawing.

TABLE III.

Spaces Numbered from beginning of Stroke.	∠r Interval of Crank from beginning of Working Stroke.	∠r Interval of Excentric Radius from Lowest Position $= \theta$.	Velocity of Piston in Inches per Second.	$\frac{\text{Piston Area.}}{\text{Porthole Area.}}$	Velocity of Flow through Portholes in Inches.	Head due to Velocity of Flow through Portholes in Feet.
	°	°	"		"	'
1	24	42	20·3	1·54	31·2	0·13
2	59	7	42·3	0·88	37·1	0·16
3	80	14	48·5	0·90	43·6	0·23
4	100	34	48·5	1·22	58·6	0·39
5	115	49	44·5	2·04	90·8	0·89
6	125	57	41·1	3·70	152·0	2·62
7	129	63	38·3	11·00	421·3	18·00

Mean head due to porthole velocity 1·30'.

TABLE IV.

Spaces Numbered from beginning of Stroke.	∠r Interval of Crank from beginning of Working Stroke.	∠r Interval of Excentric Radius from Lowest Point.	Velocity of Piston in Inches per Second.	Piston Area. Porthole Area.	Velocity of Flow through Portholes in Inches.	Head due to Velocity of Flow through Portholes in Feet.
	°	°	"		"	'
1	24	45	20·3	1·40	28·4	0·09
2	59	10	42·3	0·81	34·3	0·14
3	80	11	48·5	0·82	39·8	0·17
4	100	31	48·5	1·02	49·5	0·27
5	117	48	43·7	1·64	73·7	0·58
6	128	59	39·7	3·25	129·0	1·81
7	135	66	34·7	10·21	353·9	13·59

Mean head due to porthole velocity 1·17′.

The velocity of inlet through portholes is obtained by multiplying the mean velocity of the piston during each interval by the ratio area of piston divided by the area of the portholes corresponding with the position of the valve at the middle of each space. This ratio is equal to $\frac{\cdot 51}{\cos\theta - \cos 66^\circ}$, when the working stroke is equal to ⅝ths of the full stroke and $\frac{\cdot 51}{\cos\theta - \cos 69^\circ}$ when the former is equal to ⅞ths of the latter.

The mean theoretical heads are the nett heads due to the actual velocities. The gross heads will be equal to these divided by the coefficient of discharge through the portholes. We shall be on the safe side in adopting ·7 as the value of this coefficient, and we obtain the following values of gross heads:—

For working stroke equal ⅝ths full stroke ... ... 1·86 feet.
" " ⅞ths " ... ... 1·67 "

For different sized engines the head due to the velocity will vary as the square of the product of the number of revolutions by the length of the stroke. For similarly designed engines the length of the stroke will vary approximately as the diameter, and the number of revolutions inversely as the square root of the diameter, so that the velocity will vary simply as the diameter. We may also take as a sufficiently near approximation that the height of the middle of the working cylinder above the surface of the tail water will be equal to twice the diameter, consequently the total loss of head will be as follows. Taking head due to porthole velocity from Table III., working stroke equal to ⅝ths full stroke, the whole head lost in inches, corresponding to 10 revolutions of 3-foot cylinder engine, is equal to

$$\left(2 + \frac{2\cdot 28 + 22\cdot 32}{36}\right) \mathrm{D} = 2\cdot 7\,\mathrm{D} \text{ inches.}$$

The loss of head due to the clearance for the tail water is constant for all speeds, consequently of the whole loss 2·7 D inches, only the part ·7 D will vary with the speed. Since the head lost will vary with the square of the velocity, the total head lost for 20 revolutions, the maximum number for an engine designed for a working stroke of $\frac{3}{4}$ths the full stroke will be equal to 4·8 D inches. In order, therefore, that the efficiency of the quick-speed engine running at its maximum rate may be equal to 90 per cent. of the nett power of the water, the nett head in feet in the casing outside the cylinders must not be less than four times the diameter of the cylinder in inches. Taking head due to porthole velocity from Table IV., working stroke equal to $\frac{3}{4}$ths of full stroke, the head lost in inches, corresponding to 10 revolutions of the 3-foot cylinder engine, is equal to

$$\left(2 + \frac{2 \cdot 28 + 20 \cdot 04}{36}\right) D = 2 \cdot 6 \, D,$$

and therefore corresponding to 12 revolutions it will be equal to 2·86 inches, or in order that in this case the efficiency may be equal to 90 per cent. the nett head on the clyinder casing in feet must be nearly equal to 2½ times the diameter of the cylinder in inches. When the working stroke is equal to $\frac{9}{10}$ths of the full stroke the loss will be somewhat less than this, and corresponding to the maximum number of revolutions the efficiency will be equal to 90 per cent. of the head in the cylinder casing, if this head in feet is equal to twice the diameter of the cylinder in inches. For all velocities less than these the efficiency will exceed 90 per cent. If the length of the working stroke is shortened either by the use of link motion with two excentrics or by altering the set of one valve in the way shown on Sheet No. 1, the speed of the piston will be greater near the end of the stroke and the speed of the valve less; there will therefore be a great increase in the loss of head through wire drawing towards the close of the stroke. In the engine shown on Sheet No. 1 the minimum working stroke is equal to $\frac{7}{12}$ths of the full stroke and the alteration in the set of the valve will make porthole areas only equal to ·71 of the porthole areas for full stroke, and the loss of head due to inlet velocity will be increased in the ratio of 10 : 5. So that in the case of the 3-foot cylinder the loss of head for 10 revolutions will be somewhat greater, viz.

$$\left(2 + \frac{2 \cdot 28 + 44 \cdot 64}{36}\right) D = 3 \cdot 3 \, D \text{ inches nearly};$$

or the efficiency at short stroke compared with the efficiency at full stroke for heads in feet equal to 2½ times the diameter in inches, will be in the ratio of 87 : 90, and for heads in feet equal to 4 times, the ratio will be 88 : 90.

Corresponding to 20 revolutions, since in the above equation the loss due to velocity is equal to 1·3 D, the total loss will be equal to 7·2 D, and the efficiency at short stroke to efficiency at full stroke in the ratio 85 : 90. The corresponding ratios for a minimum stroke equal one-half the full stroke will be 81 : 90 and 86 : 90 respectively for 10 revolutions, and 80 : 90 for 20 revolutions.

This diminution of efficiency may be obviated by increasing the depth of the portholes. This with a single excentric and consequent uniform throw of a valve would involve the use of an excessively deep valve. With two excentrics, by which the length of throw would always be less than that due to the full working stroke, the diminution below 90 per cent. for a stroke equal to half full stroke may be avoided.

For a variation of the stroke from $\frac{7}{12}$ths to $\frac{3}{8}$ths, which corresponds to $\frac{1}{4}$th of the full stroke and nearly $\frac{1}{3}$rd of the working stroke, the diminution does not exceed 5 per cent. with porthole areas adapted for full stroke, therefore the quantity of water used may be economically varied by this means from about one-half the full stroke to full working stroke. For cut-offs quicker than this, the efficiency will decrease rapidly.

The diminution of the volume of water consumed may be varied at will by altering the speed of the engine. The slower the rate the greater will be the efficiency, but the variation will be trifling in amount. Leaving out of consideration this slight variation, the effort exerted by the engine is constant for all rates of work. Consequently, in order to make the engine run the desired number of revolutions, the gearing must be so adjusted to the maximum resistance to be overcome and constant effort of the engine, that the latter works at the desired speed. Let

N = constant number of revolutions of working shaft.
n_1 = maximum number of engine shaft.
n_2 = any other less number „
P = effort at piston.
L = crank shaft radius.
r_1, r_2 = radii of driving wheels of engine corresponding to the revolutions n_1, n_2.
R_1, R_2 = „ working shaft „ „ „
f_1, f_2 = efforts at periphery of engine driving wheel corresponding to radii r_1, r_2.
F_1, F_2 = efforts at periphery of working shaft wheels corresponding to radii R_1, R_2, and corresponding to constant velocity.

Then we must have

$$\frac{n_2 r_2}{R_2} = N = \frac{n_1 r_1}{R_1},$$

$$\frac{r_2}{R_2} = \frac{n_1 r_1}{n_2 R_1};$$

also

$$r_2 f_2 = r_1 f_1 = P L,$$

$$\frac{N F_2 R_2}{N F_1 R_1} = \frac{n_2 r_2 f_2}{n_1 r_1 f_1} = \frac{n_2}{n_1}.$$

The mechanical artifices for varying the speed of the pulleys to any extent in the case of shafting driven by belting is of the simplest possible character, since the machinery may be driven from either end of the engine shaft. In the case of geared wheels two separate sets may be fixed without difficulty to each end and so arranged that the act of throwing one set out of gear puts the other set into gear. By reducing the quantity of water consumed at each change by ¼th of the maximum quantity, the minimum quantity will be equal to one-half of the maximum. By reducing each time by ⅓rd the minimum quantity will be equal to ⅓rd of the maximum, and so on.

By a combination of two sets only of geared wheels—one set at each end—and link motion, the quantity of water consumed may be reduced by gradual degrees to a little over ⅓rd of the maximum without diminishing the efficiency.

The following table gives the values of the ratio of the minimum effort during a single complete revolution to the maximum corresponding to the three working strokes when acting singly and when coupled together. In the last case they must be set so that the angular interval between the crank arms of the combined machine is equal to half the angular interval of the single machine. In a single-cylinder steam engine there is a dead point of no work. The fourth column gives the value of the same ratio for cut-offs of ⅛th, ⅙th, ¼th, and ⅓rd the full stroke in a two-cylinder engine.

TABLE V.

	Working Stroke = $\frac{6}{8}$ths Full Stroke.	Working Stroke = $\frac{7}{8}$ths Full Stroke.	Working Stroke = $\frac{9}{10}$ths Full Stroke.	Double-cylinder Steam Engine.
Four-cylinder Engine ..	·46	·52	·57	·28
Six " " ..	·58	·62	·70	·34
Two fours coupled ..	·69	·73	·76	·41
Two six " ..	·72	·81	·84	·53

From the above table we see that even in the case of four-cylinder engine working singly a much less balance wheel is necessary to produce uniform motion than in the case of a double-cylinder steam engine.

Wire drawing towards the end of the stroke will make the diminution in the effort gradual.

In order to keep the speed constant for varying amounts of work a throttle valve may be inserted in the supply pipe. In the case of gases the diminution of pressure is compensated to a great extent by the increase of volume, but in the case of liquids the diminution of pressure by wire drawing through a throttle valve represents a total loss. For most purposes, however, absolute constancy of rate is not necessary so long as the change of motion is uniform. This can readily be ensured in the case of the water-pressure engine by the use of heavy fly-wheels because the effort of the engine is constant. In the case of impulse machines, such as undershot wheels and turbines, the effort on the vanes at slow speed is greater than when at best speed, so that a sudden diminution of the work would cause a sudden increase in the speed. Similarly in the case of breast wheels the buckets become more heavily loaded with water than when running at the best rate, and in consequence a sudden diminution of resistance causes a sudden increase of speed.

Tables VI., VII., and VIII. give the minimum heads corresponding to the maximum number of revolutions for 90 per cent. efficiency and the corresponding

TABLE VI.—QUICK-SPEED ENGINES.

Diameter of Cylinder in Inches.	Full Length of Cylinder in Inches.	Working Length of Cylinder in Inches.	Maximum Number of Revolutions per Minute.	Cubic Feet of Water used per Minute.	Coefficient of Efficiency = ·9. Horse-power corresponding to Minimum Heads and Maximum Number of Revolutions.										
4	16	$13\frac{1}{3}$	64	24	$\frac{3}{5}$	$\frac{9}{10}$	$1\frac{1}{5}$	$1\frac{1}{2}$	$1\frac{4}{5}$	$2\frac{1}{4}$	$2\frac{7}{10}$	3	$3\frac{3}{5}$	$4\frac{1}{2}$	$5\frac{1}{4}$
6	20	$16\frac{2}{3}$	50	53	..	2	$2\frac{2}{3}$	$3\frac{1}{3}$	4	5	6	7	8	10	12
8	25	$20\frac{5}{6}$	44	104	..	..	6	$7\frac{1}{2}$	9	11	13	16	18	22	26
10	30	25	39	170	..	..	..	$11\frac{1}{2}$	14	17	20	23	27	34	40
12	35	$29\frac{1}{6}$	36	287	..	..	..	..	23	29	35	41	47	58	69
15	44	$36\frac{2}{3}$	32	470	..	..	..	..	..	48	58	67	77	96	112
18	52	$43\frac{1}{3}$	29	730	..	..	..	..	..	..	89	104	119	148	173
21	59	$49\frac{1}{6}$	27	1040	..	..	..	..	..	..	..	143	169	211	247
24	65	$54\frac{1}{6}$	25	1380	..	..	..	..	..	..	..	..	225	281	328
30	80	$66\frac{2}{3}$	22	2340	..	..	..	..	..	..	..	..	..	476	556
36	96	80	20	3700	..	..	..	..	..	..	..	..	..	..	880
Minimum heads	..	..	..	..	16	24	32	40	48	60	72	84	96	120	140

TABLE VII.

Diameter of Cylinder in Inches.	Full Length of Cylinder in Inches.	Working Length of Cylinder in Inches.	Maximum Number of Revolutions per Minute.	Cubic Feet of Water used per Minute.	Coefficient of Efficiency = ·9. Horse-power corresponding to Minimum Heads and Maximum Number of Revolutions.										
4	16	14	40	16	$\frac{1}{4}$	$\frac{2}{5}$	$\frac{1}{2}$	$\frac{7}{10}$	$\frac{4}{5}$	1	$1\frac{1}{4}$	$1\frac{2}{5}$	$1\frac{3}{5}$	2	$2\frac{1}{2}$
6	20	$17\frac{1}{2}$	31	34	..	$\frac{9}{10}$	$1\frac{1}{5}$	$1\frac{1}{2}$	$1\frac{3}{4}$	$2\frac{1}{10}$	$2\frac{3}{5}$	3	$3\frac{1}{2}$	$4\frac{2}{5}$	$5\frac{1}{4}$
8	25	$21\frac{7}{8}$	27	67	..	..	$2\frac{1}{4}$	$2\frac{4}{5}$	$3\frac{1}{2}$	$4\frac{1}{5}$	5	6	7	$8\frac{1}{2}$	10
10	30	$26\frac{1}{4}$	24	112	..	..	..	$4\frac{3}{4}$	$5\frac{3}{4}$	7	$8\frac{1}{2}$	10	$11\frac{1}{2}$	$14\frac{1}{4}$	17
12	35	$30\frac{5}{8}$	22	180	..	..	..	..	9	$11\frac{1}{2}$	14	16	18	23	28
15	44	$38\frac{1}{2}$	20	307	..	..	..	..	..	19	23	27	31	39	47
18	52	$45\frac{1}{2}$	18	473	..	..	..	..	..	..	36	42	48	60	72
21	59	$51\frac{5}{8}$	16	648	..	..	..	..	..	..	..	57	66	82	99
24	65	$56\frac{7}{8}$	15	873	..	..	..	..	..	..	..	..	90	112	134
30	80	70	14	1570	..	..	..	..	..	..	..	..	..	200	240
36	96	84	12	2330	..	..	..	..	..	..	..	..	..	..	356
Minimum heads	..	..	..	..	10	15	20	25	30	37	45	52	60	75	90

horse-power for various sizes of four cylinder engines in which the working stroke is equal respectively to $\frac{5}{8}$ths, $\frac{7}{8}$ths, and $\frac{9}{10}$ths of the full stroke, and also the quantity of water used.

TABLE VIII.

Diameter of Cylinder in Inches.	Full Length of Cylinder in Inches.	Working Length of Cylinder in Inches.	Maximum Number of Revolutions per Minute.	Cubic Feet of Water used per Minute.	Coefficient of Efficiency = ·9. Horse-power corresponding to Minimum Heads and Maximum Number of Revolutions.										
4	16	14$\frac{2}{5}$	26	11	$\frac{1}{5}$	$\frac{1}{4}$	$\frac{3}{10}$	$\frac{2}{5}$	$\frac{1}{2}$	$\frac{3}{5}$	$\frac{7}{10}$	$\frac{4}{5}$	$\frac{9}{10}$	1$\frac{1}{5}$	1$\frac{1}{3}$
6	20	18	21	24	..	$\frac{2}{5}$	$\frac{1}{2}$	$\frac{7}{10}$	$\frac{4}{5}$	1	1$\frac{1}{4}$	1$\frac{2}{5}$	1$\frac{3}{5}$	2	2$\frac{1}{2}$
8	25	22$\frac{1}{2}$	19	48	..	..	1$\frac{3}{10}$	1$\frac{3}{5}$	2	2$\frac{1}{2}$	3	3$\frac{1}{2}$	4	5	6
10	30	27	17	81	..	..	..	2$\frac{1}{4}$	3$\frac{1}{3}$	4$\frac{1}{2}$	5	5$\frac{3}{4}$	6$\frac{1}{2}$	8$\frac{1}{4}$	10
12	35	31$\frac{1}{2}$	15	123	..	..	..	..	4$\frac{3}{4}$	6$\frac{1}{4}$	7$\frac{1}{4}$	8$\frac{1}{2}$	9$\frac{3}{4}$	12	14
15	44	40	13	208	..	..	..	..	..	10	12$\frac{3}{4}$	15	17	21$\frac{1}{4}$	25$\frac{1}{2}$
18	52	46$\frac{1}{5}$	12	320	..	..	..	..	..	..	19$\frac{1}{2}$	22$\frac{3}{4}$	26	32$\frac{1}{2}$	39
21	59	53	11	460	..	..	..	..	..	..	..	32	38	47	55
24	65	58$\frac{1}{2}$	10	618	..	..	..	..	..	..	..	..	50	63	76
30	80	72	9	1036	..	..	..	..	..	..	..	..	..	105	126
36	96	86$\frac{2}{5}$	8	1610	..	..	..	..	..	..	..	..	..	..	196
Minimum heads	..	..	..	..	8	12	16	20	24	30	36	42	48	60	72

Let K′ be the coefficient of efficiency for any height H′ less than the tabular height H, which is the minimum height corresponding to the maximum number of revolutions for which the efficiency of the machine is equal to 90 per cent. of the nett head in the cylinder casing, then

$$\frac{1 - K'}{1 - \cdot 9} = \frac{H}{H'},$$

whence

$$K' = 1 - \frac{H}{10\,H'},$$

and

$$H' = \frac{H}{10\,(1 - K')}.$$

If we assume that the coefficient of efficiency of high breast-wheels does not exceed ·7, and of turbines ·6, of the nett head, the water-pressure engine will give a higher rate of efficiency for all heights greater than those given in Table IX. for high breast-wheels, and in Table X for turbines, corresponding to the maximum number of revolutions given in Tables VI., VII., and VIII.

TABLE IX.—HIGH BREAST-WHEELS.

Diameter	″ 4	″ 6	″ 8	″ 10	″ 12	″ 15	″ 18	″ 21	″ 24	″ 30	″ 36
In Table VI... ..	5	8	11	13	16	20	24	28	32	40	48
„ VII. ..	3	5	7	8	10	12	15	17	20	25	30
„ VIII. ..	3	4	5	6	8	10	12	14	16	20	24

TABLE X.—TURBINES.

Diameter	″ 4	″ 6	″ 8	″ 10	″ 12	″ 15	″ 18	″ 21	″ 24	″ 30	″ 36
In Table VI... ..	4	6	8	10	12	15	18	21	24	30	36
„ VII. ..	3	4	5	6	7	9	11	13	15	19	22
„ VIII. ..	2	3	4	5	6	7	9	10	12	15	18

Table XI. gives the approximate sizes in feet of four cylinder vertical engines of various diameters. The height is the distance between the bottom of the casing and the centre of the crank shaft, the length of the connecting rod being assumed to be equal to six times the radius of the crank shaft. The lengths and widths represent the area covered by each machine.

TABLE XI.

Diameter	″ 4	″ 6	″ 8	″ 10	″ 12	″ 15	″ 18	″ 21	″ 24	″ 30	″ 36
Heights	7	8	10	12	14	17	20	23	26	31	38
Lengths..	3	4	5	6	7	9	10	12	16	18	19
Widths	2	2	3	4	5	6	7	8	9	10	12

Special Features of the Water-pressure Engine.

On account of the small difference between the minimum and maximum efforts on the piston, a very small balance wheel will suffice to secure uniformity of motion. When two engines are coupled together the moving parts of the machinery will probably suffice. The efficiency of the machine far exceeds that of any other water-power machine, being unquestionably fully equal to 90 per cent. of the nett head for the maximum revolutions and minimum heights given in the Tables. The less the number of revolutions and the greater the head the greater will be the efficiency, consequently any engine will work with any quantity of water less than the maximum quantity, and under all heads less than the maximum head for which the strength of the different parts of the engine have been designed.

With steam power it is impossible to keep the steam pressure in the boiler constant, therefore even where the resistance is absolutely constant no arrangement of governors can keep the rate of the engine constant. In the case of water the pressure in the cylinders is constant, and therefore when the resistance to be overcome is constant too, the motion must be absolutely uniform. For electric lighting purposes therefore, where there is no natural water-power, it may be worth while to pump the water by steam into a high reservoir in order to drive the water-pressure engine. *The great efficiency of the water-pressure engine will cause the difference in the expenditure of fuel between driving the electric machinery directly by the steam engines and intermediately by means of the water-pressure engine very slight.* By the application of the link motion the motions of the engine can be regulated like those of the steam engine. The engine can be brought to a standstill and the direction of rotation reversed. It is admirably adapted, therefore, for deep-mining purposes in mountainous countries as a winding engine. The small-sized engines admit of easy transport from place to place, so that compressed air for machine drills can be inexpensively provided even in trial mining operations. Heads of 300 feet will almost always be within reach, and for this head a supply pipe of 1½ inch diameter will convey enough water for a 1 horse-power machine, without losing more than 5 feet of head by friction in every 100 feet length of supply pipe.

No stuffing boxes are required. It will be sufficient to make all rubbing faces of gun-metal with an easy mechanical fit.

All the butt joints are planed, all the bolts turned, and all bolt holes bored, so that an intelligent labourer can put the machine together.

In these engines there is no change in the direction of the strain on the connecting rods, which are always in tension. Single brasses, therefore, only are required, which will not require adjusting, but will wear away gradually till they have to be replaced by new ones. Screws are provided for adjusting the length of the connecting rods as the brasses wear away.

In vertical engines the moving parts of the machinery exactly balance each other at every instant.

High-pressure Noiseless Quick-speed Pump.

The best combination for pumping purposes is a two-cylinder engine. In the power engine the piston-rods are always in tension whatever may be the inclination of the cylinders. In the pump the cylinders must be vertical in order that the piston-rods may be always in tension. The best shape for the cylindrical lid is that shown on the drawing. It will act as an ordinary discharge valve, giving a sufficient area of orifice, so that the noiseless slide-valve is not indispensable. The piston is a double-beat valve. The full length of the stroke is slightly in excess of the working length, so that the piston drops below the end of the working barrel and there is a short interval before the end of the stroke during which no water is passing the valve.

During this interval the valve will get quietly home on its seat, and if during very quick strokes this interval should prove too short, the valve will be forced on to its seat by the pressure in the suction pipe only, which will generally be less than the atmospheric pressure. The impact, therefore, even in this last case will be noiseless. The empty piston is brought back to its lowest position by the action of gravity. The maximum number of strokes which the pump can be run per minute will depend solely on the length of the pump barrel.

If the descent were performed in vacuum and friction were left out of consideration, the maximum number of revolutions would be given by the equation

$$n = 30\sqrt{\frac{g}{2\,\mathrm{L}}}\,; \qquad [1]$$

and since the discharge varies as the length of the stroke and the number of revolutions, it follows that for pumps of different length of cylinder and of the same diameter, the maximum discharge will vary as the square root of the length of the cylinders.

The resistance to the descent of the piston is due to two causes: (1) surface friction, (2) the difference between the pressures of the water on the two sides of the valves. As the piston and piston-rod work without packing with an easy mechanical fit in the cylinder and guide, we may leave friction out of consideration. The resistance per unit of area due to the difference of pressures will be simply due to the head producing the velocity of flow through the valve. In the case of the piston valve, we may always ensure an area of discharge equal to at least one-half the area of the cylinder. We shall therefore be on the safe side in assuming that the area of the cylinder does not exceed two and a half times the valve outlet area, and using the minimum experimental coefficient of discharge ·6, the height due to the velocity through the valve orifice will be less than $\frac{11\,v^2}{2\,g}$; to be still more on the safe side take it equal to $\frac{6\,v^2}{g}$.

If W be the weight of the valve piston and any added weight, and *w* the weight of a cubic foot of the liquid, the equation of motion neglecting friction will be

$$\frac{\mathrm{W}}{g}\cdot\frac{d^2 s}{d t^2} = \mathrm{W} - \frac{3\,\pi\,w\,\mathrm{D}^2}{2\,g}\left(\frac{d\,s}{d\,t}\right)^2 \qquad [2]$$

The maximum velocity which the piston would attain, if disconnected from the crank shaft under the resultant accelerating force given in equation [2] will be reached when the retarding force is equal to the accelerating force.

Taking as the practical limit of the number of revolutions, three-fourths of the number given by equation [1], the following table gives the maximum number of revolutions per minute corresponding to various lengths of stroke, the maximum piston speed when rigidly connected with the crank, and the maximum speed when actuated by gravity.

TABLE XII.

Length of stroke in feet	3	4	6	8
Maximum number of revolutions	50	45	36	31
„ piston speed driven by crank ..	7·8	9·4	11·2	12·9
„ „ „ actuated by gravity ..	7·2	9·0	10·4	12·0
Maximum height of middle of barrel above surface of water in well, in feet	21	14	8	3

In calculating the maximum heights of pump above surface of water in well, 2 ft., 3 ft., 4 ft., and 5 ft. have been allowed for height due to velocity in suction pipes of same diameter as pump cylinders for the lengths of stroke 3 ft., 4 ft., 6 ft., and 8 ft. respectively. For slow speeds the depths may of course in each case be equal to over 20 feet.

The pump will work with equal efficiency when buried over head in water.

The table shows that when the pumps are being driven at the quickest rate the velocity of the piston at middle stroke, if it were rigidly connected with the crank, would be greater than it is when it descends under the action of gravity only. From this it follows that the piston rope will be slack during the middle part of the down stroke, and will not become tight till the piston passes the point at which its velocity, if it were rigidly connected with the crank, would be less than the maximum velocity due to the action of gravity.

The instantaneous impact on the rope will be simply equal to the momentum of the weight of the valve and its attached weight due to the difference between the velocity acquired under the action of gravity and the velocity the piston would have had at the instant of tightening of the rope, if it had been rigidly connected with the crank. This momentum will therefore in all cases be very small.

The strain on the piston-rods being always a tensile strain, a sudden sucking in of the air through the emptying of the well will not cause any damage to the pumping machinery. In fact, there will scarcely be any difference in the strain on the rope, because the opening of the delivery valve will admit the water from the rising main and at once produce the maximum pressure due to the lift on the confined air, so that the steadiness and rate of motion of the machine would not be perceptibly interfered with.

On the return stroke the difference between the maximum and minimum strain will be equal to the maximum pressure on the piston due to the weight of the column of water raised increased by the height due to velocity in the rising main and the head necessary to overcome friction, &c. In the case of new ropes, therefore, there will be considerable amount of extension, if they have not previously been subjected to the necessary proof load before they are put in use. In the case of the most seasoned ropes there will probably be an appreciable amount of stretching. If the size of the steel ropes are chosen so that the maximum strain per square inch does not exceed 12,000 lb. to the square inch, and we adopt 30,000,000 lb. as the modulus of elasticity for steel in extension, the increase in length for every hundred feet length of rope would not exceed half an inch.

The stretching of the rope, therefore, would interfere with the efficiency of the pump, because the piston under pressure would not rise to the top of the working barrel. The greater the length of the barrel, therefore, the less will the efficiency of the pump be affected by this stretching of the rope. If the barrel be 4 feet long the extension will amount to less than 1 per cent. of the length of the pump barrel for every hundred feet of rope. The weight of the fuel required to raise a given volume of water would not be in the least affected by the extension of the rope, because within the working limits of stress no permanent set will be produced in the rope, and the work done in stretching the rope at the commencement of the stroke before the piston begins to rise will be given back again on the return stroke before the piston begins to descend. When the balanced valve is in use, the direction of rotation of the pump shaft must be always the same. When the direction of the rotation is reversed the water will be let into the cylinder during the down stroke and the machine will act as a power engine, doing fully 90 per cent. duty. There are no air-valves to the lid, the contained water will therefore not be discharged from the cylinder whilst the piston is passing the lowest point, but will be pumped back again into the rising main. Since the other half of the machine would be simultaneously acting as a power engine, the engine power required to keep up the motion would be less than 20 per cent. of the whole work done. If sniffcocks are provided in connection with the pump barrel, as shown on the drawing Sheet No. 2, the snore-box holes may be at any time flushed out by opening the sniffcock tap and shutting off the steam. The water in the lift would of itself reverse the direction of rotation until the whole of it had been discharged back again through the snore box into the pump. These sniffcocks may also be used for supplying air to the air-vessel. No injury therefore will be done to the machinery by any accidental reversing of the direction of rotation. When the same engine is used for both winding and pumping, the balanced valve must be suspended in front of the portholes. The water will then be discharged entirely through the openings formed above the cylinder by the raising of the lid, which will act simply as a valve. If the piston were rigidly connected with the crank-arm, the return of the valve to its seat would be instantaneous and the force of the impact due to the pressure of the whole head of water in the rising main upon the whole area of the lid. The action would therefore be too violent to admit of quick speed. As, however, the principle of action of this pump admits of the use of an extensible rope, the return of the lid valve to its seat would take place before there was any sensible difference between the pressures in the water above and below the lid, and the force which would bring back the valve to its seat would probably not exceed that due to its own weight and the pressure of the water on the difference of the areas of the upper and under sides of the lid, except for rates of speed approaching the maximum rate admissible with the use of the balanced cylindrical valve. The following table gives full particulars of pumps with cylinders varying from 2 inches to 36 inches in diameter and lengths varying from 8 inches to 8 feet working length. The actual lengths of the cylinders must be from 1 to 3 inches more, to admit of the descent of the piston below the cylinder.

Column I. gives the load in tons on a crank-arm 1 foot long, which will produce the same amount of torsion in the shaft as that produced by the statical load on the piston with a crank-arm equal to half the length of the pump barrel.

Column II. the quantity of water delivered per minute in gallons.

TABLE XIII.

Diameter of Cylinder in Inches.	Statical Load on Piston due to 100 Feet Lift in tons.	Working Stroke 8 Inches.	Number of Strokes per Minute 100.	Working Stroke 12 Inches.	Number of Strokes per Minute 90.	Working Stroke 18 Inches.	Number of Strokes per Minute 70.	Working Stroke 2 Feet.	Number of Strokes per Minute 62.	Working Stroke 3 feet.	Number of Strokes per Minute 50.	Working Stroke 4 Feet.	Number of Strokes per Minute 45.	Working Stroke 6 feet.	Number of Strokes per Minute 36.	Working Stroke 8 feet.	Number of Strokes per Minute 31.
		I.	II.	I.	II.	I.	II.	I.	II.	I.	II.	I.	II.	I.	II.	I.	II.
2	0·06	0·02	18	0·04	24	0·05	28	..	..	..	..	..	..	..	..	..	..
3	0·13	..	..	0·07	54	0·1	63	0·14	75	..	..	..	..	..	..	..	..
4	0·23	..	..	0·13	96	0·2	110	0·25	130	0·4	160	..	..	..	..	..	..
6	0·5	..	..	..	..	0·4	250	0·54	300	0·8	360	1·04	430	..	..	..	..
8	1·0	..	..	..	..	..	..	1·1	520	1·6	640	2·1	770	3·1	920	..	..
10	1·5	..	..	..	..	..	..	1·6	820	2·3	1,000	3·2	1,200	4·7	1,500	6·2	1,700
12	2·1	..	..	..	..	..	..	..	..	3·4	1,440	4·6	1,700	6·5	2,000	8·6	2,300
15	3·3	..	..	..	..	..	..	..	..	5·3	2,250	6·9	2,700	10·2	3,300	13·5	3,700
18	4·7	..	..	..	..	..	..	..	..	..	..	10·0	3,900	14·6	4,600	19·3	5,300
21	6·4	..	..	..	.	..	..	..	..	..	..	13·4	5,300	19·8	6,300	25·2	7,400
24	8·4	..	..	..	..	..	..	..	..	..	..	17·4	7,000	26·0	8,300	34·5	9,500
30	13·2	..	..	..	..	..	..	..	..	..	..	..	..	41·0	13,000	54·0	14,900
36	19·0	..	..	..	..	..	..	..	..	..	..	..	..	59·0	18,700	78·0	21,400

Sheet No. 2 contains the detail drawings of a pump with cylinders 8 inches diameter and 4 feet working stroke, designed for deep-pit pumping.

An union screw capable of shortening the rope about 2 feet is provided for the exact adjustment of the length. For the regulation of the stretching during the first use a chain of length proportioned to the depth of the pit must form part of the cable. This may be shortened link by link till the rope becomes seasoned to the work. Two notches are cut in the sides of the piston-rod which are opposite the edges of the lubricators when the piston is at the top and bottom of the stroke. The length of the ropes ought to be adjusted when the piston is at the bottom of the stroke free from the pressure in the lift. A diaphragm, which forms part of the suction pipe, will prevent the piston from dropping out of the cylinder, if the adjustment of the rope is neglected. The only effect, therefore, of this neglect would be a diminution of the pump efficiency, owing to the diminished travel of the piston.

In the drawing the ascending piston is just about to commence the working stroke and the returning piston has descended two inches. In order to ensure the effectual commencement of the working stroke before the porthole opens, the valve may be set so that it has to descend nearly one and a half inches before the porthole opens after the working stroke has begun. Whilst the valve is descending this interval the piston, leaving extension of rope out of consideration, will have risen nearly one inch. During this interval the water will be forced through the lid valve. The lagging of the valve at the beginning of the stroke is of course accompanied by a similar one at the end of the stroke, so that the water flows back into the cylinder at the commencement of the down stroke. Since, however, in the case of the beginning of the descent the piston is at dead point, the space through which it will descend whilst the valve, then travelling at its maximum speed, closes up the one and a half inch opening will be less than two-thirds of an inch in the 4-foot stroke. This space is too small to effect the efficiency of the pump, but the non-closing of the porthole before the commencement of the return stroke effectually prevents the disturbance of the lid valve on its seat, and so ensures the noiseless action of the pump. The loss of efficiency of the whole machine will be inappreciable, because the descending water will be doing work.

This admissibility of the valve lagging behind the piston without interfering with the efficient working of the pump is a most important feature, because an error of adjustment of the slide-valve up and down of full one and a half inch will likewise not affect the efficiency of working, because the top of the porthole will not be opened before the upward working stroke begins, and the lap of the valve at the bottom being more than one and a half inch, the portholes will not open at the bottom when the valve is at its highest point.

At the middle stroke of the piston the valve will be travelling at its slowest speed, so that although the instant of maximum opening will be slightly delayed, the diminution of the depth of the porthole at the instant the piston is at mid-stroke will not exceed one-eighth of an inch.

The pump cylinder does not act as a guide for the balanced valve, which moves freely up and down under the action of gravity, the centre of gravity being situated in the centre of the valve rod, which in addition works in a guide 8 inches long.

The two pumps are perfectly independent, so that one can be kept at work whilst the other is undergoing examination. A sluice valve with slow motion to separate the lift from the pump is not necessary, because the pressure on both sides of the valve is the same so long as the pump case is full of water, and the valve may easily therefore be pushed into position to close the hole. In order to prevent any possibility of accident through neglect to open the sluice before the pumping commences, the sluice lid itself should contain a self-acting valve, or be itself, as shown in the drawing, simply a flap valve. This sluice can therefore remain in position until the pump has been set to work and the pressure on both sides equalised. This equalisation of the pressure will prevent any sensible impact of the valve on its seat.

The contents of the air-vessel shown on the drawing are equal to about eight times the contents of each cylinder, so that a regular discharge with almost constant velocity will be maintained in the rising main. This last is shown to be 6 inches in diameter throughout, but it may be increased to any size on leaving the air-vessel.

At maximum speed the two pumps will throw nearly 800 gallons a minute. This discharge corresponds to an average velocity in a 6-inch pipe about 11 feet a second. The loss by friction in 100 feet of pipe would be about 6 feet, or the maximum increase due to friction would amount to about 6 per cent. of the statical load on the piston due to the height of the lift. With a rising main of the same diameter as the pump barrel the maximum increase would be equal to less than 2 per cent. of the statical load.

The vertical pump has the same special features as those enumerated for the water-pressure engine, and compared with other pumps has another special feature which makes it invaluable for deep-pit pumping. In the case of ordinary pumping-engines at starting the air drawn through the foot valve from the suction pipe has on the return stroke to be forced through a reflux valve either in the piston itself or in the rising main. When the rising main is empty, the pressure necessary to open the valve must exceed that due to the weight of the valve by the amount of the atmospheric pressure on the difference of the areas or the absolute pressure must considerably exceed the atmospheric pressure, and when the main is full by this difference and the whole pressure of the water in the main, which on a 400-foot lift would amount to more than 160 lb. to the square inch. In the pump at present under consideration, the discharge valve is actuated independently of the motion of the fluids pumped, and the upper face of the valve fixed in the piston is during the whole descent exposed to a pressure less than that of the atmosphere; so that the absolute pressure on the under side of the valve necessary to raise the valve will be less than the atmospheric pressure.

Again when the pump is started after examination with the rising main full of water, the air drawn from the suction pipe is not forced through the reflux valve at the bottom of the column of water, but into the space between the casing and the pump cylinder. In the pump shown on Sketch No. 2, the contents of this external case are equal to sixteen times the contents of one foot length of pump cylinder, so that to take an extreme case of a depth of 16 feet between the top of the pump barrel and the surface of the water in the well, the water would have risen through the piston valves before the absolute pressure on the upper side reached 28 lb. to the square inch.

It is this feature of the present pump that allows of the use of a piston working in the cylinder with an easy mechanical fit without packing of any sort. There may be a slight forcing through of the fluids pumped, not only between the piston and cylinder and valve and cylinder, but also between the piston and valve rods and the guides in which they act. In the former case the leakage cannot be perceived externally, and the loss due to this cause will be very much less than the loss due to the work necessary to overcome packing friction. In order to prevent the escape of water from interfering with the oiling of the piston-rods, two oil-cups are provided in the guide brasses, one below the other. In the lower one there is a clear annular space round the piston about half an inch deep. Any liquid, therefore, which may be forced through between the piston and the guides does not reach the upper oil-box but escapes over the edges of the lower, in the case of the engine through the air-pipes back to the cylinder, in the case of the pumps directly into the sump.

Two or three small annular nicks cut in the sides of the piston valve and guides will in all probability, however, prevent any perceptible leakage.

The high speeds claimed for this pump are admissible for three reasons :—

(1) The strain on the brasses is always in the same direction.

(2) Owing to wire-drawing of the water as the piston forces itself into the working barrel, and the elasticity of the rope, the strain at the commencement of the stroke is not brought on instantaneously.

(3) The masses of the ropes and the pistons are the only ones subject to a change in direction of momentum. As the weights of one set of ropes and the attached piston and its weight rarely reach 15 cwt., this change of momentum is very trifling, is nearly wholly effected by gravity, and for reason (1) could not in any case cause a sudden "shock."

In the case of lifts actuated by beams the usual length of stroke is about 6 feet, and the arms of the beams about 9 feet. The beam with its three arms usually weighs about 5 tons, and its motion, so far as change of momentum is concerned, may be considered equal to that of an ordinary fly-wheel of the same weight and about 12 feet diameter which is revolving at about one-third the rate of the engine fly-wheel. This change in the direction of rotation occurs twice in each stroke. The rods of the lifts may be taken to weigh a ton for every 100 feet in length. In the case of a single lift this weight may be sufficient to keep the strain on the brasses in tension, but the change in momentum during the second half of the down stroke and the first half of the up stroke is produced wholly by the engine, and is the same in intensity as if the masses of the rope and the piston were fixed to the end of the 9-feet arm. The work done would be equivalent to changing the direction of momentum of a 9 ton fly-wheel of about 15 feet diameter. Since the piston is forced down by the weight of the rods, if any joint is loose each change of direction must be accompanied by a shock to the machinery. In the case of a double lift the change of momentum in the weight of the rods will be partially effected throughout by gravitation, but the change in momentum in the beams will of course require the expenditure of twice as much work. Leaving out of consideration, however, the masses of the beams and the rods, there must be a change in the direction of the force exerted at whatever speed the engine may be going. The impact of this is instantaneous, and if there be the least play in the brasses or in the joints of the rods there must be a shock equivalent to the blow given by a monkey, whose weight is equal to the pressure on the piston, and which has fallen through a height due to the velocity of impact. Since this height varies as the square of the velocity of impact, the violence of the shock would vary as the square of the number of revolutions per minute. With a 16-inch diameter lift 400 feet deep the weight of the monkey would be fully 13 tons.

No footstep valve is required, if the adjustment of the rope is attended to with ordinary care. Without a footstep valve, the valves in the piston at the commencement of the downstroke will not open until the slide valve, purposely set to lag behind the piston, closes the portholes. With a foot valve they will be opened the instant the piston begins to descend, and the footstep valve will be forced on to its seat under the maximum pressure in the rising main, and consequently close during quick speed with a violent impact, if some special means of gently closing the valve are not provided. The annular air space shown on Sheet No. 2, in the design for a cast iron barrel, answers this purpose admirably. *During the ascent of the piston, the pressure of the air will be less than that of the atmosphere, and at the commencement of the return stroke the valve will also be returned to its seat with a pressure less than that due to the atmosphere, whenever the surface of the water in the well is below the pump barrel.*

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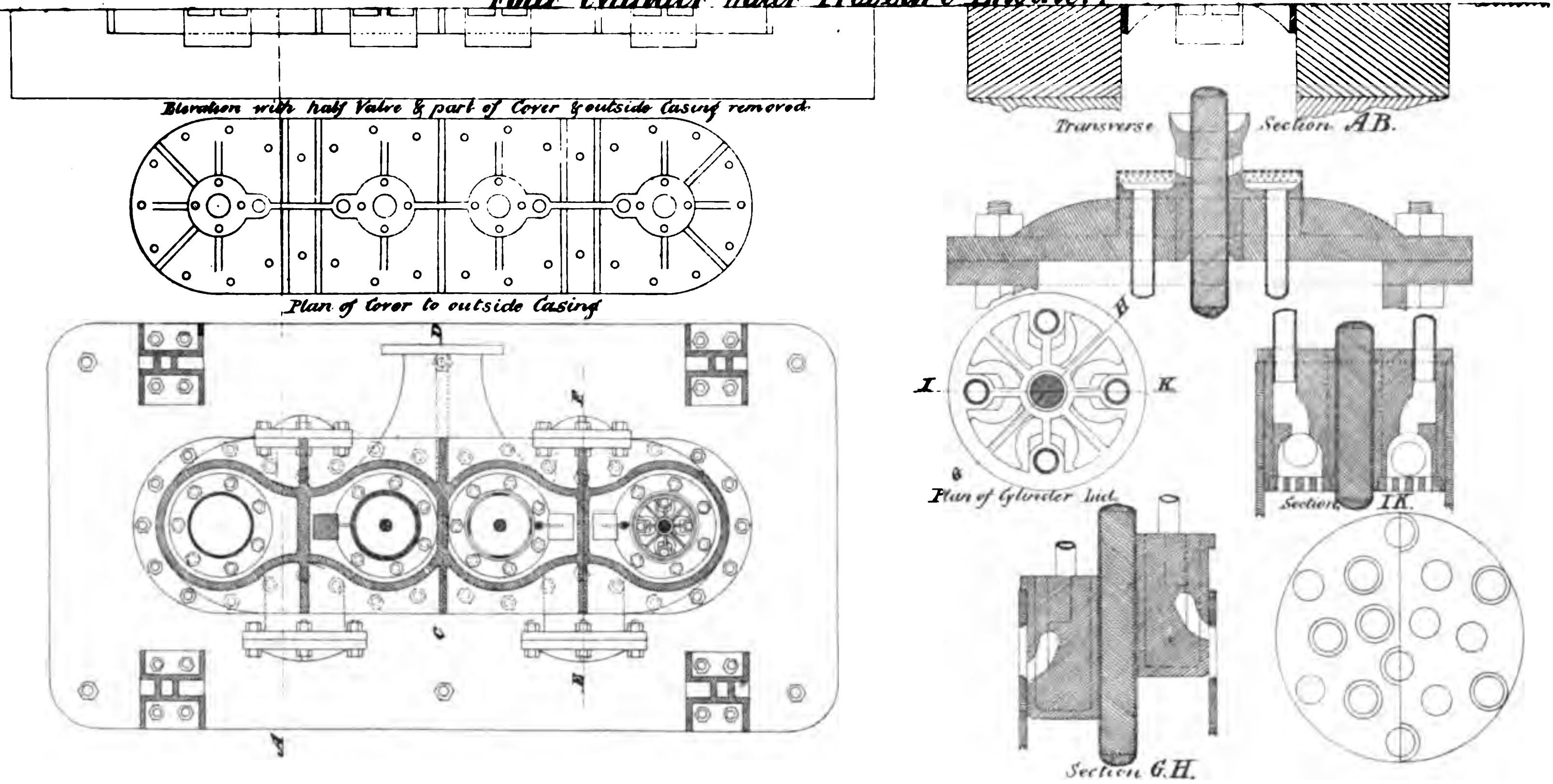
Sheet No. 1.

Four Cylinder Water Pressure Engine.


Elevation with half Valve & part of Cover & outside Casing removed.
Plan of Cover to outside Casing
Transverse Section AB.
H
I
K
Plan of Cylinder Ind.
Section IK.
Section GH.

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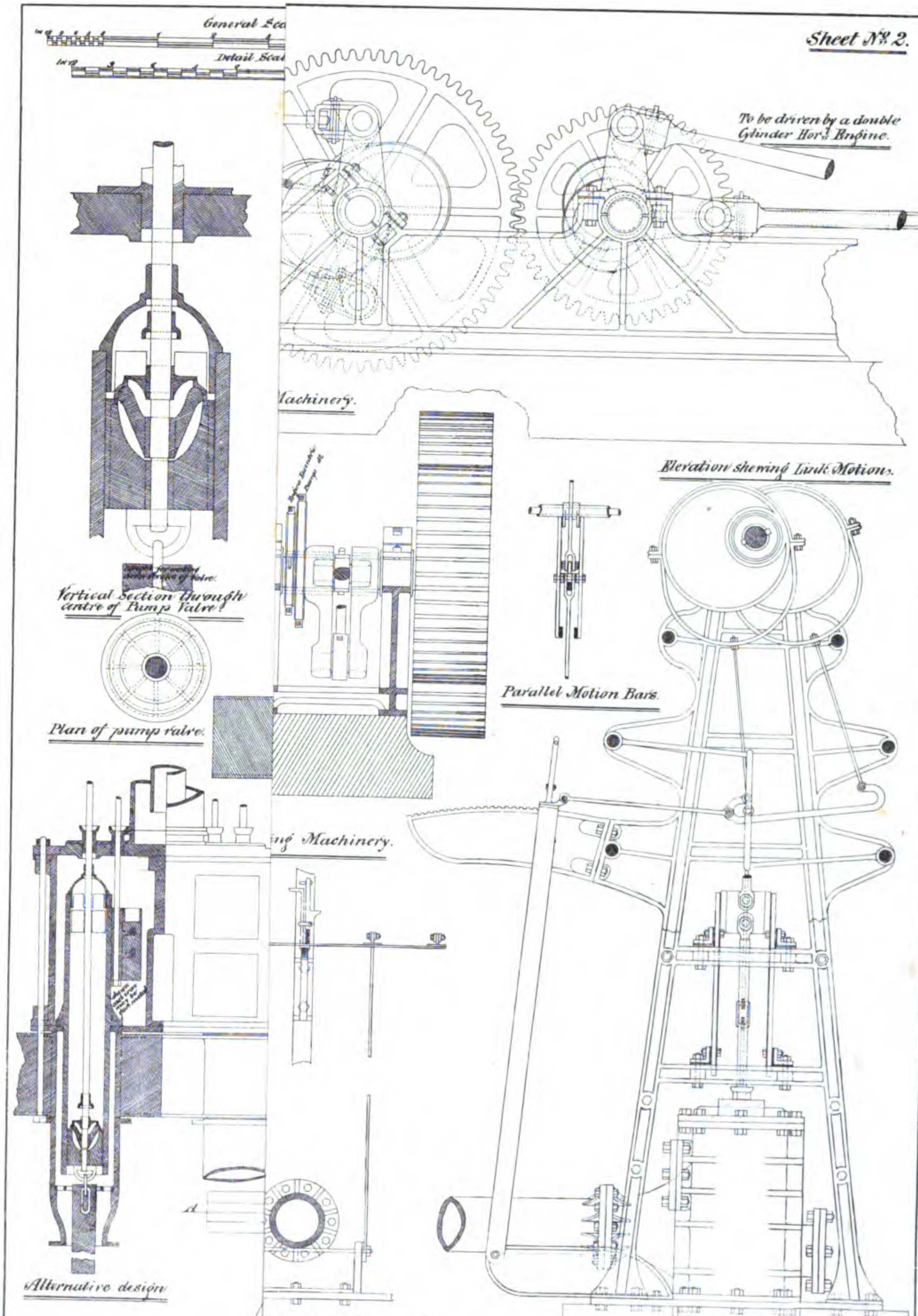
Sheet No. 2.



achinery.

Vertical Section through centre of Pump Valve.

Plan of pump valve.

Parallel Motion Bars.

Elevation shewing Link Motion.

ing Machinery.

Alternative design

Motion adjusting Lever.

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7'